NASA Technical Memorandum 86988

Correlation of Rheological Characteristics of Lubricants with Transmission Efficiency Measurements

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Prepared for the
International Tribology Conference
sponored by the Japan Society of Lubrication Engineers
Tokyo, Japan, July 8-10, 1985

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E-2531

SUMMARY

The power efficiency of a helicopter transmission has been analyzed for 11 lubricants by looking at the Newtonian and non-Newtonian properties of the lubricants. A non-Newtonian property of the lubricants was the limiting shear strength proportionality constant. The tests were performed on a high-pressure, short-time shear strength analyzer. The Newtonian and non-Newtonian properties of the lubricants were used in obtaining the following formula for the power efficiency, where η_0 is in pascal-seconds and α is per gigapascals: $E = 1 + 1.93\,\eta_0\alpha^{-3.81} - 0.282\,\gamma$

INTRODUCTION

Mitchell and Coy (1982) presents results from the efficiency testing of 11 lubricants in the main rotor transmission of an OH-58 helicopter. The efficiency ranges from 98.3 to 98.8 percent, depending on the lubricant used. Furthermore, with two exceptions, the efficiency for a given lubricant increases with increasing oil inlet temperature. This generally high efficiency was not surprising since it has long been recognized that the mechanical efficiency of helicopter power trains is quite high. Usually a planetary reduction has 3/4 percent loss, and a single bevel or spur gear mesh has 1/2 percent loss, as pointed out by Shipley (1962). Compared with the large amounts of power available from the engines of a helicopter, it may seem that fractions of a percent of power lost in the power train path are inconsequential. However, the higher the losses, the larger and heavier the oil cooling systems required. This effect contributes to lowering both helicopter payload and survivability. Moreover, in light of Mitchell and Coy's (1982) finding of as much as 50-percent variation in power loss among the lubricants they tested, the effect on oil cooling system weight, size, and vulnerability became very real. By proper selection of a lubricant the operating envelope and payload capacity of a helicopter can be improved.

Many factors act together in causing the power loss in a helicopter transmission, which is a rather complicated assembly of gears, shafts, seals, and bearings. In a typical application sliding, windage, churning, and pumping losses all play a role, as do the rheological lubricant properties. In their tests Mitchell and Coy varied only the lubricant. Their results were repeatable, and they took great care to ensure that the system was flushed clean

before a new lubricant was used. They also varied the order of testing. Therefore it could be concluded that the power losses observed from their tests are entirely due to the different lubricants used.

Our objective was to see if, by defining the rheological properties of the lubricant, we could obtain a better analysis of the power efficiency results. The test results from Mitchell and Coy (1982) are presented herein, but details of the test apparatus and test procedure are not repeated. For these details, consult Mitchell and Coy (1982). Once the results are presented, the various lubricant properties are discussed. As observed by Mitchell and Coy, using just the viscosity of the lubricant to describe its rheological behavior does not give an understanding of the results. Additional properties of the fluid can be divided into two groups, the Newtonian and non-Newtonian properties. The Newtonian properties of the various lubricants are described in Present, et al. (1983). We evaluated the minimum film thickness in the lubrication conjunction from the Newtonian properties of the lubricants. We determined their non-Newtonian properties in a test apparatus developed by Jacobson (1985). An efficiency formula that comprises both the Newtonian and non-Newtonian properties of the lubricant was developed.

SYMBOLS

E	efficiency
$\tilde{E}$	efficiency as obtained from equation (5)
E_1	$(E - \tilde{E})100/E$
e	modulus of elasticity, Pa
e'	effective elastic modulus, $2\bigg/\left(\dfrac{1 - \nu_a^2}{e_a} + \dfrac{1 - \nu_b^2}{e_b}\right)$
F	applied normal load, N
G	dimensionless materials parameter, $\alpha e'$
H	dimensionless film thickness, h/R_x
H_{min}	dimensionless minimum film thickness, h_{min}/R_x
h	film thickness, m
h_{min}	minimum film thickness, m
k	ellipticity parameter, $(R_y/R_x)^{2/\pi}$
p	pressure, Pa
p_s	solidification pressure, Pa
R_x	effective radius in x-direction, m

R_y effective radius in y-direction, m

r curvature radius, m

U dimensionless speed parameter, $u\eta_0/e'R_x$

u mean surface velocity, $(u_a + u_b)/2$, m/s

W dimensionless load parameter, $F/e'R_x^2$

α pressure-viscosity coefficient of lubricant, Pa^{-1}

β temperature-viscosity coefficient of lubricant, 1/C°

γ limiting shear strength proportionality constant

η viscosity, Pa-s

η_k kinematic viscosity, η/ρ, m^2/s

η_0 lubricant viscosity measured at inlet temperature, Pa-s

θ impact angle

θ_0 critical impact angle

ν Poisson's ratio

ρ lubricant density, kg/m^3

τ shear stress, Pa

τ_L limiting shear strength, Pa

τ_0 shear strength at atmospheric pressure, Pa

Subscripts:

a solid a

b solid b

HELICOPTER TRANSMISSION RESULTS

All of the lubricants tested by Mitchell and Coy (1982) were near the 5- to 7-cS range in lubricant kinematic viscosity and were qualified for use in helicopter transmissions. The test lubricant types are shown in table I. The experimentally determined efficiencies are plotted against oil inlet temperature in figure 1. The efficiencies ranged from 98.3 to 98.8 percent. This is an overall variation in losses of almost 50 percent relative to the losses associated with the maximum efficiency measured.

In general, a higher test temperature for a given lubricant yielded a higher efficiency (fig. 1). The exceptions were with lubricants E and C,

which have quite different kinematic viscosities. Lubricant G, being more viscous, could not be tested at the targeted oil inlet temperature. The two automatic transmission fluids (A and B) and the type I synthetic gear lubricant (E) yielded significantly lower efficiencies as a group.

LUBRICANT RHEOLOGICAL PARAMETERS

The gears and rolling-element bearings used in helicopter transmissions operate at oil film pressures in the lubricated conjunction of 1 to 3 GPa. At these high pressures the mineral oils convert to a solid (amorphous) state and yield a shear strength of the same order as that of a soft steel. It therefore was concluded that defining the lubricant just in terms of the viscosity is not sufficient and that other parameters that better characterize performance in these nonconformal contacts are necessary.

The following five fluid rheological parameters were arrived at as characterizing the lubricant used in the gears and rolling bearings in helicopter transmissions:

(1) Dynamic viscosity of lubricant at atmospheric pressure, η_0, Pa-s

(2) Pressure-viscosity coefficient of lubricant, α, Pa^{-1}

(3) Temperature-viscosity coefficient of lubricant, β, 1/°C

(4) Limiting shear strength proportionality constant, γ, dimensionless

(5) Solidification pressure, or pressure at which lubricant changes from liquid to solid, p_s, Pa

The first three parameters define the Newtonian behavior of the fluid, and the last two define its non-Newtonian behavior.

ELASTOHYDRODYNAMIC LUBRICATION FILM THICKNESS

The gears and rolling-element bearings used in helicopter transmissions are normally lubricated elastohydrodynamically. The Newtonian properties of the lubricant (η_0, α, and β) are used to describe the minimum film thickness in the conjunction. The minimum film thickness formula of Hamrock and Dowson (1977) is used. It is

$$H_{min} = 3.63\ U^{0.68} G^{0.49} W^{-0.073} (1 - e^{-0.68k}) \qquad (1)$$

In this equation the dominant exponent occurs on the speed parameter, while the exponent on the load parameter is very small and negative. Maintaining a fluid-film thickness of adequate magnitude is clearly vital to the efficient operation of helicopter transmissions.

In the helicopter transmission tests only the lubricant was changed so that this effect as it relates to equation (1) is

$$h_{min} = c\eta_k^{0.68}\alpha^{0.49} \tag{2}$$

where c contains the parameters of equation (1) other than those relating to the lubricant and η_k is the kinematic viscosity of the lubricant η/ρ in square meters per second. The values of η_k and α for the 11 lubricants at three temperatures were obtained from Present et al. (1983) and are shown in tables II and III.

Figure 2 is based on the information of tables II and III and equation (2). It shows that, if lubricants G and H are excluded, there are two main groups of lubricants. The group forming thick films is A, E, and B, and the group forming thinner films is F, D, C, K, I, and J. These groupings coincide with the efficiency results in figure 1. That is, the thicker film group (A, B, and E) is the low-efficiency group in figure 1. Furthermore, the thinner film group C, D, F, I, and J corresponds to high-efficiency group in figure 1.

The fluids G and H do not fit into either group in figure 2 because their kinematic viscosities are quite different from the others, G being much more viscous and H much less viscous.

NON-NEWTONIAN LUBRICANT PARAMETERS

Lubricants follow two distinct types of shear behavior. At low shear stresses most lubricants behave as Newtonian liquids (i.e., the shear stress is directly proportional to the shear strain rate). At high shear stresses most lubricants behave like plastic solids (i.e., the shear stress is a function of the lubricant type, the pressure, and the temperature and is independent of the lubricant viscosity).

The great severity of the lubrication conditions in hard elastohydrodynamic contacts has called into question the normal assumption of Newtonian behavior of the lubricant. Jacobson and Hamrock (1984) redefines the pressure and mass flow rate equations depending on how the values of shear stress at the surface compare with the limiting shear stress of the lubricant. The limiting shear strength is linearly dependent on pressure, or

$$\tau_L = \tau_0 + \gamma p \tag{4}$$

where

τ_0 shear strength at zero pressure
τ_L limiting shear strength
γ limiting shear strength proportionality constant

The limiting shear strength proportionality constant γ can also be seen as the slope of the limiting shear strength - pressure relationship $\partial\tau_L/\partial p$.

Figure 3 describes the difference between a Newtonian and non-Newtonian lubricant model. The fluid model is Newtonian except when the shear stress reaches the limiting shear strength value. At this point the shear stress is set equal to the limiting shear strength. The value of γ is a function of temperature. The limiting shear strength proportionality constant is a important parameter used in describing the non-Newtonian behavior of lubricants.

APPARATUS TO MEASURE γ

Jacobson (1985) describes the development of a simple high-pressure, short-time shear strength analyzer (figs. 4 and 5). The main parts of the apparatus are a spherical ball and two parallel flat surfaces. The ball is placed on a sledge (fig. 5) that rests on two inclined guides. When the sledge slides along the guides due to the acceleration of gravity, the ball does not rotate. At the bottom of the guides is a lubricated, polished, flat horizontal surface made of cemented carbide. The polished steel ball hits the lubricated flat surface with an inclination determined by the inclination of the guides. At the start of the impact the ball does not rotate, but during the impact the shear stresses from the oil acting on the steel ball accelerate a rotational velocity of the ball. If the shear strength of the lubricant film is high enough, the ball will accelerate until it moves in pure rolling at the end of the impact. After the impact the ball falls onto the hard, horizontal top surface of a cart. The cart moves easily in the direction of the motion of the ball. The cart motion is measured by using strain gauges and a digital oscilloscope.

When the impact angle θ between the vertical direction and the inclined guides is decreased, the momentum transferred to the cart is also decreased, as shown in figure 6, until it reaches zero at the angle θ_0. This critical impact angle determines the limiting shear strength proportionality constant γ. Jacobson (1985) establishes the relationship between θ_0 and γ to be $\gamma = \tan \theta_0/7$.

POWER EFFICIENCY FORMULA

Table IV shows the values of γ for the 11 transmission fluids as obtained by using the approach of Jacobson (1985). Also shown are the Newtonian properties of the fluids as obtained from Present et al. (1983) as well as the power efficiency of the fluids (new fluids assumed) as obtained from Mitchell et al. (1982). For this table it is assumed that the temperature is 100 °C and the pressure is atmospheric. Making use of this information while exploring a number of different forms of equations led to the discovery that the best fit of the lubricant parameters with the power efficiency was

$$\tilde{E} = 1 + 1.93\,\eta_0\alpha^{-3.81} - 0.282\,\gamma \tag{5}$$

where the viscosity η_0 is expressed in megapascals and α per gigapascal. The second term on the right side of equation (5) describes the Newtonian contribution of the lubricant to the power efficiency; the third term represents the non-Newtonian contribution. The non-Newtonian contribution is generally one order of magnitude larger than the Newtonian contribution. Also shown in table IV is the power efficiency as obtained from equation (5) and the percentage difference between it and the power efficiency obtained from Mitchell and Coy (1982). This percentage difference is defined as

$$E_1 = \frac{(E - \tilde{E})100}{E} \tag{6}$$

The values of E_1 in table IV are within ±0.15 percent, which is extremely good.

COMMENTS ON LUBRICANTS A, B, C, AND E

A result of Mitchell and Coy (1982) that has not been mentioned is the effect of temperature on power efficiency for lubricants C and E of figure 1 and the lower efficiencies for lubricants A, B, and E. In figure 1 we see that for oils C and E the slope of efficiency versus temperature has the opposite sign from that obtained for the other oils tested. To describe this non-Newtonian effect, the work of Höglund (1984) will be introduced. The essential difference between the Höglund apparatus and the Jacobson (1985) apparatus is that the Jacobson apparatus measures γ dynamically to 7.5 GPa at room temperature, while the Höglund apparatus measures γ and the solidification pressure p_s to 2.2 GPa and 200 °C. Höglund's measurements were made for a broad sample of lubricants that included many of the types used in the present study. He found the solidification pressure to increase with temperature. In Höglund's study the ranking of lubricants with increasing solidification pressure at 100 °C was as follows: a synthetic traction fluid (1.07 GPa), a lithium soap grease (1.37 GPa), three paraffinic mineral oils (1.48, 1.66, 1.77 GPa), and finally the synthetic hydrocarbon and synthetic ester lubricants, which did not solidify to the limit of the test rig (2.2 GPa) at 100 °C. These synthetics did solidify at lower temperatures, in one case as low as 40 °C for a polyalphaolefin/polyolester synthetic lubricant.

What is significant about Höglund's results in relation to the present study is that he shows there can be a large difference in manifested frictional losses among various lubricants at the same pressures and temperatures as a result of the solidification pressures being different. We believe that the lower efficiencies with lubricants A, B, and E are due to a lower solidification pressure for these lubricants combined with a higher γ.

Kuss, et al. (1983) shows that adding 9.6 percent sulfur to a base stock drastically changes the viscosity-pressure characteristics. The addition of sulfur produces a knee in the viscosity-pressure relation beyond which the viscosity increases even more rapidly with pressure. The measured viscosities presented in tables II and IV and figure 2 are for atmospheric pressure only. The viscosity at the high pressures in the contact regions of gears and rolling-element bearings would be different from that calculated by using the pressure-viscosity coefficients with an exponential relation.

The reason for the decrease in efficiency with increasing in temperature for lubricants C and E is unknown, but we speculate that, instead of the bearings and gears in the transmission being lubricated elastohydrodynamically, they may be lubricated by boundary or mixed lubrication, where asperity contacts occur. Another possibility is that the slope reversal of lubricants C and E may be due to the increased activity of the particular additive packages at the higher temperatures. Present et al. (1983) reports lubricant E as having large amounts of chlorine, zinc, sulfur, and barium, which indicate the presence of large amounts of antiwear and detergent additives.

CONCLUSIONS

The power efficiency results of a helicopter transmission for 11 lubricants have been analyzed by looking at the Newtonian and non-Newtonian properties of these fluids. A reasonable correlation between minimum film thickness and power efficiency was found as long as the viscosities of the lubricants were similar.

The limiting shear strength proportionality constants for the 11 fluids were measured on a high-pressure, short-time shear strength analyzer. The Newtonian and non-Newtonian properties of the lubricants were used in obtaining a formula for the power efficiency.

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TABLE I. - TEST LUBRICANT TYPES

[From Mitchell and Coy (1982).]

Lubricant	NASA code (AFLRL code)	Specification	General type	Base stock[a]
A	11252	Dexron II GM 6137-M	Automatic transmission fluid	Mineral oil
B	11268	Dexron II GM 6137-M	Automatic transmission fluid	Mineral oil
C	11250	MIL-L-23699	Turbine engine oil	Ester (PE)
D	11254	MIL-L-23699	Type II synthetic gas turbine oil	Ester (PE)
E	11256	-------------------	Formulated gear lubricant	Dibasic acid ester
F	11258	-------------------	NASA gear test lubricant - synthetic paraffinic with antiwear additives	Synthetic hydrocarbon (PAO)
G	11260	MIL-L-2104C MIL-L-46152	Synthetic fleet engine oil	Mixture of 80 percent synthetic hydrocarbons (PAO) and 20 percent ester (TMP)
H	11262	MIL-L-7808	Turbine engine oil	Ester (TMP)
I	11264	MIL-L-23699	Type II turbine engine oil	Mixture of 50 percent TMP ester and 50 percent PE ester
J	11270	MIL-L-23699	Type II turbine engine oil	Ester (PE)
K	11266	-------------------	Turbine engine oil	Mixture of 99 percent PE ester and 1 percent DPE ester

[a]PE = pentaerythritol; TMP = trimethylolpropane; PAO = polyalphaolefin; and DPE = dipentaerythritol.

TABLE II. - KINEMATIC VISCOSITY DATA ACCORDING TO ANSI/ASTM SPECIFICATION D-455

[From Present et al. (1983).]

Lubricant code	Kinematic viscosity at listed temperature, η_k, cS		
	40 °C	60 °C	100 °C
A	37.48	10.48	7.01
B	33.15	9.64	6.52
C	26.40	7.69	5.13
D	26.17	7.50	5.00
E	33.91	8.91	5.87
F	28.01	8.15	5.36
G	56.65	15.05	9.83
H	13.16	4.73	3.38
I	24.19	7.18	4.85
J	24.76	7.23	4.89
K	26.39	7.61	5.09

TABLE III. - PRESSURE-VISCOSITY COEFFICIENTS FOR TEST LUBRICANTS

[From Present et al. (1983).]

Lubricant code	Pressure-viscosity coefficient at listed temperature, α, GPa^{-1}		
	40 °C	60 °C	100 °C
A	15.37	11.72	10.22
B	14.96	11.85	10.34
C	11.63	10.03	8.81
D	12.43	9.94	8.71
E	15.53	11.51	9.88
F	13.44	11.14	9.53
G	13.80	11.34	10.36
H	11.53	9.14	7.95
I	12.08	9.24	8.34
J	11.96	9.23	8.30
K	11.40	9.50	8.32

TABLE IV. - POWER EFFICIENCY AND RHEOLOGICAL CHARACTERISTICS OF LUBRICANTS

Lubricant code	Dynamic viscosity at 100 °C and 1 atm, mPa-s	Pressure-viscosity coefficient,[a] α, GPa^{-1}	Shear strength proportionality constant, γ	Power efficiency of new oil,[b] E	Power efficiency,[c] $\tilde{E}$	$E_1 = \frac{(E - \tilde{E})100}{E}$, percent
A	5.94	10.73	0.0588	0.9840	0.9848	-0.081
B	5.56	10.84	.0598	.9833	.9844	- .112
C	4.89	9.85	.0514	.9876	.9871	.051
D	4.80	9.72	.0523	.9860	.9869	- .091
E	5.33	11.53	.0603	.9835	.9836	- .041
F	4.23	10.85	.0543	.9865	.9856	.091
G	8.34	10.34	.0571	.9873	.9861	.122
H	2.97	8.79	.0563	.9870	.9856	.142
I	4.63	8.95	.0570	.9864	.9860	.041
J	4.57	8.95	.0527	.9864	.9872	- .081
K	5.34	8.65	.0543	.9869	.9875	- .061

[a]From Present et al. (1983).
[b]From Mitchell and Coy (1982).
[c]From equation (5).

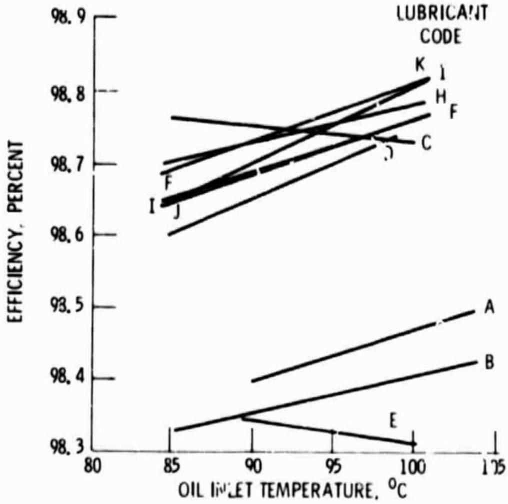

Figure 1. - Experimental efficiency correlated with oil inlet temperature.

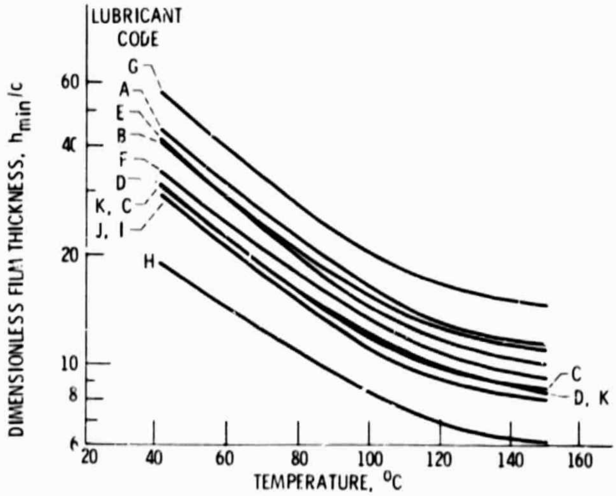

Figure 2. - Effect of dimensionless film thickness on temperature for 11 lubricants.

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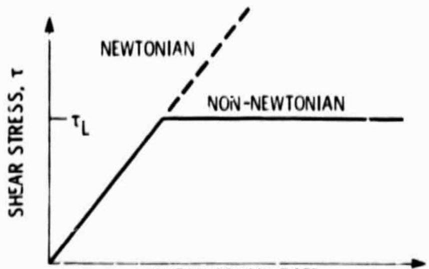

Figure 3. - Lubricant model.

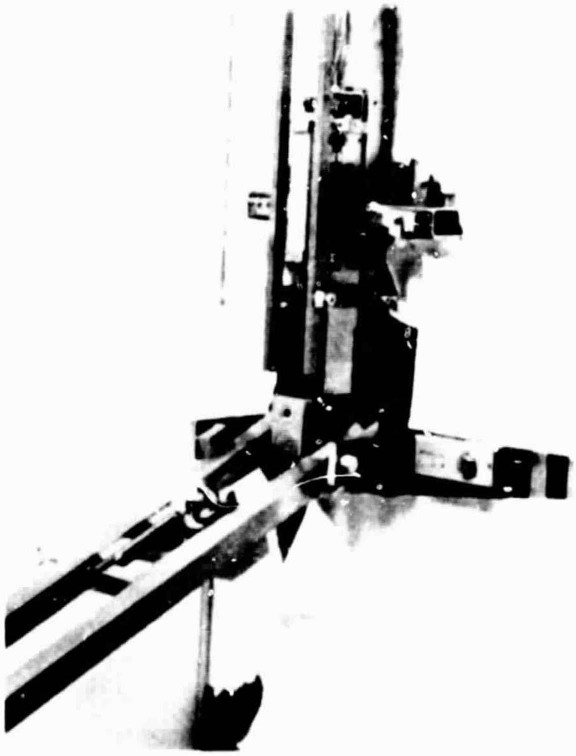
Figure 4. - Test apparatus.

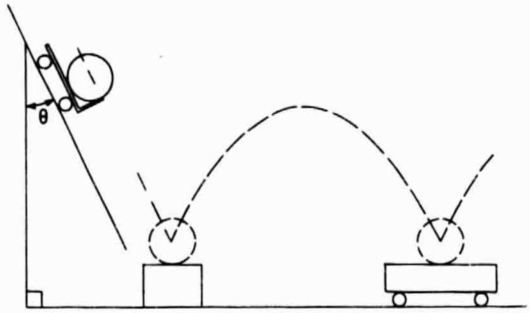

Figure 5. - Sketch of test apparatus.

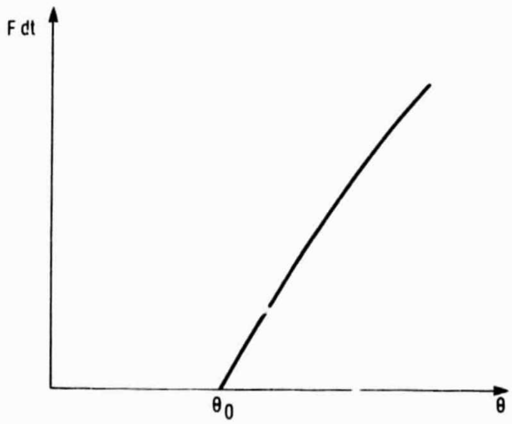

Figure 6. - Momentum transferred to cart as a function of impact angle θ (see fig. 5). The critical angle θ_0, when $\tan \theta_0 / 7 = \gamma$, is indicated.

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA TM-86988		

4. Title and Subtitle	5. Report Date
Correlation of Rheological Characteristics of Lubricants with Transmission Efficiency Measurements	
	6. Performing Organization Code: 505-33-62

7. Author(s)	8. Performing Organization Report No.
Bo O. Jacobson, Bernard J. Hamrock, and Erik Höglund	E-2531
	10. Work Unit No.

9. Performing Organization Name and Address	11. Contract or Grant No.
National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135	
	13. Type of Report and Period Covered: Technical Memorandum
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546	14. Sponsoring Agency Code

15. Supplementary Notes

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16. Abstract

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17. Key Words (Suggested by Author(s))	18. Distribution Statement
Rheology; Lubricant; Helicopter transmissions; Elastohydrodynamic lubrication	Unclassified - unlimited STAR Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of pages	22. Price*
Unclassified	Unclassified		

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